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A REPORT ON THE MAIN VALVE BEARING REBUILD FOR THE ADVANCED TURBINE AEROTHERMAL RESEARCH RIG (ATARR)

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On December 16, 1992 the main valve in the ATARR facility seized during a run at 50 psia and 400F. Immediately prior to this problem, a run at 90 psia and 400F was successfully completed. These two tests were the first ATARR runs at high temperature. The main valve had been run dozens of times prior to this at room temperature without incident. However, some wear of the soft nickel coating on the cylinder was observed before the high temperature runs.

This section of the main valve final report documents the process of rebuilding the main valve and includes a description of an analyses conducted to determine the possible cause or causes of the failure. The bearing redesigned process, which was based on the analysis just noted, is detailed. The actual rebuild, final inspection and testing of the main valve are also included to complete this section.

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1.0 INTRODUCTION - MAIN VALVE REBUILD

On December 16, 1992 the main valve in the ATARR facility seized during a run at 50 psia and 400 °F. Immediately prior to this problem, a run at 90 psia and 400 °F was successfully completed. These two tests were the first ATARR runs at high temperature. The main valve had been run dozens of times prior to this at room temperature without incident. However, some wear of the soft nickel coating on the cylinder was observed before the high temperature runs.

This section of the main valve final report documents the process of rebuilding the main valve and includes a description of an analysis conducted to determine the possible cause or causes of the failure. The bearing redesigned process, which was based on the analysis just noted, is detailed. The actual rebuild, final inspection and testing of the main valve are also included to complete this section.

2.0 FAILURE ANALYSIS

There are three possible causes for the main valve failure. All three of which probably contributed to the failure. The first possibility is that a variation in the thermal expansion rate among the valve components caused a reduction in the bearing clearance. This is most likely to have been the major contributor to the failure. Another possibility is that the slider became misaligned due to uneven thermal expansion and seized on the cylinder. The previous bearing configuration could not tolerate any misalignment of the bearings on the cylinder. The last possibility considered is that the bearing and/or cylinder surfaces became galled to the point of seizing. The soft nickel plating on the cylinder was observed, during the room temperature experiments, to be susceptible to galling.

The first step in the failure investigation was the disassembly and inspection of the main valve. The valve cylinder and bearings were inspected by the WPAFB zone shop on 1/5/93. The cylinder's dimensions are given in Table 1 and the bearing's dimensions are given in Table 2. The results show that the cylinder outside diameter was out of tolerance at 10 of the 24 measurement locations. The free end of the cylinder is out of round by 0.005 in. and most of the length is out by 0.004 in.. The bearing inside diameters were also out of tolerance. The clearance between the cylinder and bearing at room temperature could have ranged from as low as 0.001 to as high as 0.011 in., but no interference is apparent.

Table 1 - Cylinder O.D. Measurements Taken 1/5/93

Drawing dimension: 16.9944 - 6.9960 in.

Axial Position	Radial Position	Dimension (in.)
1.5	3-9	16.996
	12-6	16.995
	4-11	16.995
5.5	3-9	16.996
	12-6	16.994
	4-11	16.995
9.5	3-9	16.996
	12-6	16.993
	4-11	16.995
13.5	3-9	16.995
	12-6	16.993
	4-11	16.997
17.5	3-9	16.996
	12-6	16.994
	4-11	16.998
21.5	3-9	16.996
	12-6	16.994
	4-11	16.998
24.5	3-9	16.997
	12-6	16.994
	4-11	16.998
27.5	3-9	16.993
	12-6	16.998
	4-11	16.997

Table 2 - Bearing I.D. Measurements Taken 1/5/93

Drawing dimension: 17.0000 - 17.0026 in.

Bearing Location	Face	Dimension (in.)
front	front	16.999
front	rear	17.002
rear	front	17.003
rear	rear	17.004

A visual inspection of the main valve components revealed that the bearings and cylinder surfaces were heavily scored. The gray cast iron bearings were scraped and the cylinder was sent out for rework to West Milton. West Milton turned down the cylinder

outside diameter to 16.982 in. to remove all of the nickel plating and scored parent metal. The cylinder O.D. was then brought back into specification with a 13 mil plating of hard chromium.

To determine what role the thermal expansion of the valve assembly played in the failure, an experimental investigation was designed to measure the thermal expansion of all the major valve components. The valve slider (5143CM-075) and reworked cylinder (5143CM-083) were sent to Calspan ATC the first week in March, 1993. The investigation procedure is outlined below. This investigation provided the coefficients of expansion for the components, information on the critical dimensions, and uneven thermal expansion of the parts.

Thermal Expansion Test Procedure

1.) Clean both slider and cylinder.

2.) Inspect the following dimensions at room temperature and record the parts surface temperature.

- Slider Valve, 5143CM-075:
- 18.1250 / 18.1275, Diameters -B- & -C-: measure each diameter at 6 locations, 3 at struts and 3 in between struts. Record the location of each measurement.
- Perpendicularity of Diameters -B- & -C- with Surface -A-.
- True Position of Diameter -C- with Diameter -B-.
- Cylinder, Slide Valve Support, 5143CM-083:
- 16.9944 / 16.9960 Diameter: measure the diameter at 3 equally spaced axial locations along diameter. At each axial location measure the diameter at 3 circumferential locations 120 deg. apart. Record the location of each measurement.

3.) Instrument both Slider and Cylinder with thermocouples.

- Slider Valve, 5143CM-075: Place 3 thermocouples equally spaced, 120 deg. apart, on both Diameter -B- & -C-. Also place 3 thermocouples equally spaced on outer skirt.

- Cylinder, Slide Valve Support, 5143CM-083: Place 9 thermocouples on 16.9960 / 16.9944 Diameter, one every 120 deg. at each of the 3 axial measurement locations.

- Make thermocouple leads long enough to extend out of the furnace to monitor the part temperatures.

4.) Heat Slider and Cylinder to 100, 200, 300 and 400 deg. F and repeat the inspection outlined in procedure 2 for each temperature, except for the following measurements that should be done only at 400 deg. F. Record the actual temperature of each measurement location as the measurement is being taken.

- <u>Slider Valve, 5143CM-075:</u>

- Perpendicularity of Diameters -B- & -C- with Surface -A-.

- True Position of Diameter -C- with Diameter -B-.

The results of the room temperature dimensional inspection are given in Tables 3 and 4. Table 3 gives the slider valve dimensions which show a 0.003 out-of-roundness of Dia. C and 0.001 out-of-roundness of Dia. B. The schematic of the slider valve in Figure 1 shows the measurement locations. Both diameters B and C are slightly out of tolerance in their perpendicularity with Datum A, 0.0030 for B and 0.0015 for C. Diameter C exceeds its true position tolerance with Dia. B by 0.002. These deviations are minor and will not adversely affect the part's performance. Table 4 gives the cylinder results which show no deviation in the reworked outer diameter.

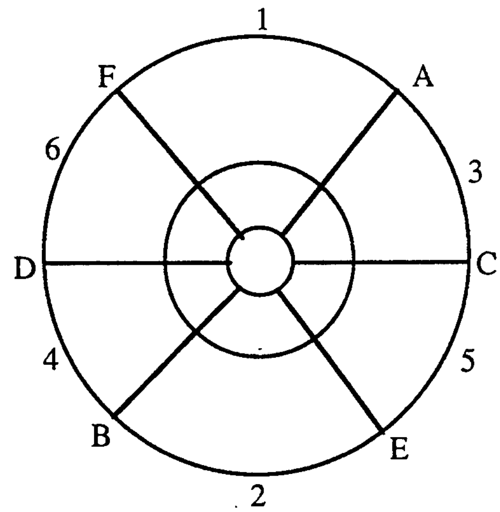


Figure 1 - Slider Measurement Locations

Table 3 - Slider Valve, P/N 5143CM-075, Inspection Results

B.) Drawing dimension: 18.1275 - 18.1250

Dia C Loc	Dimension (in.)	Dia B Loc	Dimension (in.)
1-2	18.1250	1-2	18.1250
3-4	18.1240	3-4	18.1260
5-6	18.1270	5-6	18.1255
A-B	18.1260	A-B	18.1265
C-D	18.1260	C-D	18.1260
E-F	18.1260	E-F	18.1260

Table 3, Continued - Slider Valve, P/N 5143CM-075, Inspection Results

B.) Drawing dimension: Dia B & C Perpendicular to Surface A within 0.002 in.

Dia C Loc	FIR (in.)	Dia B Loc	FIR (in.)
1	+0.0015	1	+0.0015
2	+0.0020	2	+0.0020
3	+0.0000	3	+0.0030
4	+0.0035	4	+0.0050
5	+0.0000	5	+0.0035
6	+0.0025	6	+0.0000
A	+0.0010	A	+0.0025
B	+0.0025	B	+0.0015
C	+0.0000	C	+0.0035
D	+0.0030	D	+0.0000
E	+0.0010	E	+0.0035
F	+0.0020	F	+0.0010

C.) Drawing dimension: Dia C True Pos. with Dia. B within 0.002 in.

Dia C Loc	FIR (in.)
1	+0.0035
A	+0.0015
3	+0.0005
C	-0.0005
5	-0.0020
E	-0.0010
2	-0.0000
B	+0.0015
4	+0.0035
D	+0.0040
6	+0.0040
F	+0.0040

FIR = Full Indicator Reading

Table 4 - Cylinder, P/N 5143CM-083, Inspection Results

Drawing dimension: Diameter 16.9969 - 16.9944 in.

Axial Loc	Cir. Loc	Dimension (in)
front	1-1	16.995
	2-2	16.995
	3-3	16.995
middle	1-1	16.995
	2-2	16.995
	3-3	16.995
back	1-1	16.995
	2-2	16.995
	3-3	16.995

back = flanged end

The thermal expansion results are given in terms of an overall average thermal coefficient of expansion for each component. The data from the 100 °F point was discarded because of the large uncertainty associated with measuring such small expansions. The remaining data were statistically analyzed and a 95% confidence interval is provided for each mean coefficient. This confidence interval was calculated from Equation 1.

$$\bar{x} - t_\beta \frac{s}{\sqrt{n}} \leq \mu \leq \bar{x} + t_\beta \frac{s}{\sqrt{n}} \tag{1}$$

where, $\bar{x}$ is the sample mean, s is the sample standard deviation, n is the sample size, t_β is the t distribution factor at the desired percent confidence, $(1-2\beta)100$, and μ is the mean of the population.

A few weeks after the slider and cylinder tests, one of the gray cast iron bearings used in the initial installation was tested to determine its coefficient of thermal expansion. The bearing expansion was measured at 275 and 375 deg. F, and provides the last piece of information in the thermal expansion puzzle.

Slider Valve, 5143CM-075:

$$\alpha = 6.389 \times 10^{-6} \pm 0.172 \times 10^{-6} \frac{1}{°F}, \text{ where n=36}$$

Cylinder, 5143CM-083:

$$\alpha = 7.202 \times 10^{-6} \pm 0.140 \times 10^{-6} \frac{1}{°F}, \text{ where n=27}$$

Bearing, Gray Cast Iron, 5143CM-078:

$$\alpha = 6.023 \times 10^{-6} \pm 0.677 \times 10^{-6} \frac{1}{°F}, \text{ where n=6}$$

Distortion of the slider valve at elevated temperature due to uneven growth was a concern because it could have caused the slider and bearings to bind on the cylinder. The measurement of the bearing containment diameters' (B and C) location and orientation at elevated temperature was designed to investigate this possibility. The results shown in Table 5 indicate very little distortion or uneven expansion. The perpendicularity of Diameter B is out of tolerance only by 0.001 and Diameter C is out by 0.002. This is an overall improvement from the measurements at room temperature. The true position of Dia. C is out of tolerance by 0.004 which is up from the 0.002 out at room temperature.

Table 5 - Slider Valve, P/N 5143CM-075, Inspection Results at 200 deg. F

B.) Drawing dimension: Dia B & C Perpendicular to Surface A within 0.002 in.

Dia C Loc	FIR (in.)	Dia B Loc	FIR (in.)
1	-0.0005	1	+0.0000
2	+0.0000	2	+0.0000
3	-0.0030	3	+0.0010
4	+0.0020	4	-0.0020
5	-0.0030	5	+0.0020
6	+0.0005	6	-0.0030
A	-0.0025	A	+0.0000
B	+0.0000	B	-0.0010
C	-0.0040	C	+0.0010
D	+0.0015	D	-0.0030
E	-0.0020	E	+0.0020
F	+0.0000	F	-0.0020

C.) Drawing dimension: Dia C True Pos. with Dia. B within 0.002 in.

Dia C Loc	FIR (in.)
1	+0.0000
A	-0.0060
3	-0.0020
C	-0.0055
5	-0.0005
E	-0.0050
2	-0.0000
B	-0.0050
4	+0.0030
D	-0.0035
6	+0.0010
F	-0.0030

FIR = Full Indicator Reading

The results from the thermal expansion tests clearly suggest that an incompatibility exists in the expansion rates of the major valve components for the initial configuration. The following analysis was conducted to determine exactly what the consequences of this

incompatibility could have been. The bearing clearance at the 400 °F would have been as follows.

Expansion of slider valve bearing containment diameters B and C:

$$\Delta D = 18.126(400-72)(6.4x10^{-6}) = 0.038 \text{ in.}$$

Expansion of the gray cast iron bearings inner diameter:

$$\Delta D = 17.001(400-72)(6.0x10^{-6}) = 0.033 \text{ in.}$$

Expansion of the cylinder outside diameter:

$$\Delta D = 16.995(400-72)(7.2x10^{-6}) = 0.040 \text{ in.}$$

The bearing clearances as determined from the measurements taken on 1/5/93:

Room Temperature: +0.001 to +0.011

At 400 °F: -0.006 to +0.004

This analysis suggests the possibility of a significant interference fit between the bearings and cylinder. Note that the slider expands more than the bearings, allowing the bearings to lose their positive seating in the slider. This could have allowed them to become misaligned, further reducing the bearing clearance.

One can conclude from this analysis that the variation in the thermal expansion rate among the components was the primary cause for the December 16th failure. One can also speculate that the bearing configuration's inability to tolerate misalignment and the galling of the soft nickel platting on the cylinder were contributing factors in the failure.

3.0 BEARING REDESIGN

The bearing redesign had to address each of the three possible causes for the failure. The soft nickel platting had already been replaced with hard chromium at the time of this redesign effort. That left the thermal expansion of the various components and the bearing misalignment issues.

The thermal expansion issue was complicated by the fact that the cylinder's coefficient of thermal expansion was larger than that of the slider's. The only viable solution was to choose a bearing material with a thermal coefficient of expansion large enough to expand the slider bearing containment rings to maintain adequate clearance at elevated temperatures. This required that a constrained thermal expansion analysis be conducted for the bearing and slider assembly. Figure 2 gives a schematic of this statistically indeterminate assembly.

<u>Constrained Thermal Expansion Analysis</u>

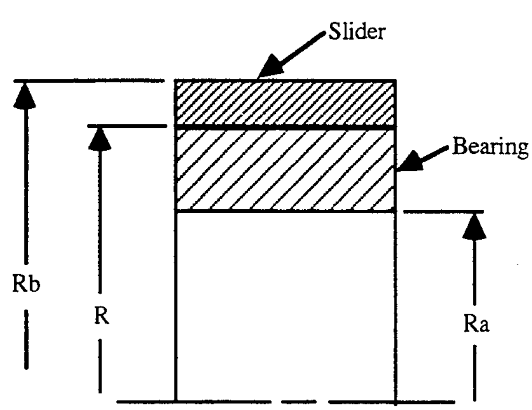


Figure 2 - Schematic of slider/bearing assembly

The radial displacement of this assembly at elevated temperature was determined by considering both the displacement due to thermal expansion and the displacement due to the interference force between the two parts. Knowing that the parts remain in contact, the sum of the displacements for each component were set equal to formulate a compatibility equation. The analysis is outlined below.

Assumptions

1.) Materials are homogeneous with uniform properties.

2.) Thermal coefficient of expansion for the bearing material (α_B) is greater than that for the slider material (α_S).

3.) Stress due to interference between parts can be modeled as a hoop stress.

Displacements due to thermal expansion

$$\delta_{S,T} = \alpha_S(\Delta T)R \tag{2a}$$

$$\delta_{B,T} = \alpha_B(\Delta T)R \tag{2b}$$

Displacements due to force between parts

Hoop stress and corresponding strain:

$$\sigma = \frac{PR}{t} \tag{3a}$$

$$\varepsilon = \frac{PR}{tE} \tag{3b}$$

writing the thickness, t, in terms of the radius and multiply by R to obtain the displacement at the interface,

$$\delta_{S,F} = \frac{PR^2}{E_S(R_b - R)} \tag{4a}$$

$$\delta_{B,F} = \frac{PR^2}{E_B(R - R_a)} \tag{4b}$$

Compatibility Equation

Formulating the compatibility equation knowing that the sum of each component's displacements must be equal to the others.

$$\delta = \delta_{S,T}+\delta_{S,F} = \delta_{B,T}-\delta_{B,F} \tag{5}$$

Making the appropriate substitutions and solving for the pressure (P) and radial displacement (δ).

$$\delta = \alpha_S(\Delta T)R+\frac{PR^2}{E_S(R_b-R)} = \alpha_B(\Delta T)R-\frac{PR^2}{E_B(R-R_a)} \tag{6}$$

$$P = \frac{\Delta T(\alpha_B-\alpha_S)E_B E_S(R-R_a)(R_b-R)}{R[E_B(R-R_a)+E_S(R_b-R)]} \tag{7}$$

$$\delta = R(\Delta T)\left\{\alpha_B - \frac{(\alpha_B-\alpha_S)E_S(R_b-R)}{E_B(R-R_a)+E_S(R_b-R)}\right\} \tag{8}$$

Material Selection

The next step in the bearing redesign was the selection of a material which would meet the thermal expansion criterion and have the high compressive strength and fatigue characteristics necessary for this bearing application. The slider/bearing assembly must match or slightly exceed the thermal expansion of the cylinder; therefore, the thermal expansion coefficient of the cylinder was used along with Equation 8 to test all candidate materials. Rearranging Equation 8 and setting the left hand side equal to the cylinder's thermal coefficient of expansion,

$$\alpha_C = \frac{\delta}{R(\Delta T)} \leq \alpha_B - \frac{(\alpha_B-\alpha_S)E_S(R_b-R)}{E_B(R-R_a)+E_S(R_b-R)} \tag{9}$$

The best candidate found for the bearing material was the copper alloy C95400. This is a well established bearing material with excellent compressive strength and fatigue resistance in its precipitation heat treated state. The pertinent material specifications are summarized below.

Copper Alloy C95400, Heat Treated

Tensile Strength: 105 ksi

Yield Strength (.5% Extension Under Load): 54 ksi

Elongation in 2 inches: 8%

Hardness (Brinell-3000 kg): 195

Compressive Strength (0.1 in. set/in.): 120 ksi

Fatigue Strength (100 million cycles): 35 ksi

Modulus of Elasticity: 15,500 ksi

Thermal Coefficient of Expansion: 9.0E-6 1/F

Analyzing this copper alloy with its properties along with the properties and dimensions of the slider in Equation 9 yields

$$\alpha_C = 7.202\text{x}10^{-6} \leq \alpha_{S/B} = 7.65\text{x}10^{-6} \tag{10}$$

where, $\alpha_S = 6.4\text{x}10^{-6} \frac{1}{^\circ F}$, $E_S = 3.0\text{x}10^7$ psi , $R_b = 9.375$ in., R=9.063 in., $R_a = 8.500$ in.

Equation 10 shows that the C95400 copper alloy meets the thermal expansion criterion with a margin of safety which is desirable considering that the analysis does not take into account any added stiffness due to the slider's six struts. A small increase (>0.005) in the bearing diametric clearance is desirable at high temperature because of the possibility of uneven growth.

The thermal coefficient of expansion of three C95400 test samples from different foundries was measured at Calspan. This added step was taken to verify the coefficient because of its importance in this application. The foundries do not monitor this property and rarely measure it. The expansion of the samples was measured at 200, 300, 400 and 500 °F. The average measured coefficients for all three samples are approximately 5%

greater than the value quoted as shown below. There is an estimated 4% uncertainty associated with these coefficient values.

Vendor	α 1/F
Advanced Bronze	9.53E-6
Farmer's Copper	9.57E-6
Arrow Metals	9.48E-6

Advanced bronze of Lodi Ohio was chosen because of a combination of price and delivery. Approximately twice the amount of required material for the two bearings was ordered so that the excess material could be used to make a replacement set of bearings when necessary.

Bearing Configuration Changes

The last issue to address in the bearing redesign is the possible misalignment and binding of the bearings on the cylinder. It is very difficult to eliminate the possibility of misalignment by dimensional control. The slider tolerances are already very tight and the part is as well machined as can be expected for a weldment of its size. The best approach is to desensitize the assembly to misalignment by redesigning the bearings. Theoretically, the slider/bearing assembly rests on the cylinder in a two point contact; top dead center of each bearing. Any attempt to provide additional surface area for contact would over constrain the slider and possibly cause it to bind, which is the case in the previous design. To provide only the necessary two points of contact, the bearing inside diameter faces were given a large modified radius as shown in Figure 3.

The level of compressive stress in the bearings at the single point of contact was a source of concern with this new bearing configuration. An applicable bearing stress analysis was found in *Roark's Formulas for Stress & Stain* and is presented here. Figure

4 is a duplication of the chart used in the analysis. It shows that the bearing and cylinder are modeled as two cylinders in contact with their axes at right angles. This is a conservative model because the bearing in a sense is a cylinder which raps completely around the other cylinder increasing its area of contact and reducing the stresses proportionally.

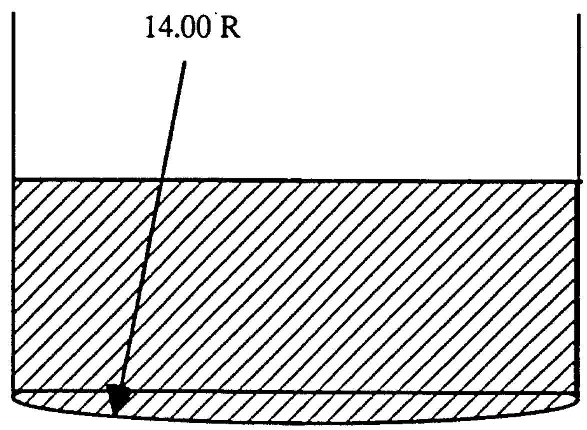
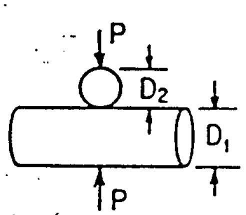


Figure 3 - Cross Section Showing Bearing Face Radius

Cylinder on a cylinder; axes at right angles



$$c = \alpha \sqrt[3]{PK_D C_E}$$
$$d = \beta \sqrt[3]{PK_D C_E}$$
$$\text{Max } \sigma_c = \frac{1.5P}{\pi c d}$$
$$y = \lambda \sqrt[3]{\frac{P^2 C_E^2}{K_D}}$$
$$\text{Max } \tau \approx \tfrac{1}{3}(\text{max } \sigma_c)$$

$K_D = \frac{D_1 D_2}{D_1 + D_2}$ and α, β, and λ depend upon $\frac{D_1}{D_2}$ as shown

D_1/D_2	1	1.5	2	3	4	6	10
α	0.908	1.045	1.158	1.350	1.505	1.767	2.175
β	0.908	0.799	0.732	0.651	0.602	0.544	0.481
λ	0.825	0.818	0.804	0.774	0.747	0.702	0.641

Figure 4 - Roark's Chart for Bearing Stress Analysis

The following variable definitions apply in the bearing stress analysis:

c - major semiaxis of elliptical contact area

d - minor semiaxis of elliptical contact area

$$C_E = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}$$

The following variable values were used to determine the bearing stress:

$D_1 = 16.995$ in. $\quad D_2 = 28.00$ in. $\quad K_D = 10.576 \quad \frac{D_1}{D_2} = 1$

$P = 250$ lbf $\quad \upsilon_1 = \upsilon_2 = 0.3 \quad C_E = 8.904\text{x}10^{-8}\ \frac{\text{in}^2}{\text{lbf}}$

The contact area and resulting compressive stress is given below:

$c = d = 0.056$ in $\quad \sigma_c = 38.0$ ksi

The predicted bearing compressive stress is a factor of three below the maximum allowable compressive strength of the C95400 bearing alloy.

The bearing redesign was finalized by custom fitting the bearing outside and inside diameters to the existing slider and cylinder, respectively. The bearing inside diameter was set by the current cylinder outside diameter plus a minimum of 0.0040 diametric clearance with the same Belcan tolerance of 0.0025.

Bearing inside diameter dimensions:

low limit = 16.9955 + 0.0040 = 16.9995

high limit = 16.9995 + 0.0025 = 17.0020

The bearing outside diameter was set by the current slider minimum and maximum diameters and the original -0.0008 to +0.0033 Belcan assembly fit.

Bearing outside diameter dimensions:

low limit = 18.1270 - 0.0033 ~ 18.1235

high limit = 18.1240 - (-0.0008) ~ 18.1250

The bearing dimensional and material changes were consolidated in a new bearing drawing by Calspan designated as H109-0039. A copy of this drawing is given as an Appendix to this report.

4.0 BEARING REBUILD

Two new bearings were machined in accordance with the H109-0039 drawing. The bearings were assembled into the slider and their inside diameters were measured. The fit between the bearings and the slider was an ideal light press fit. The results of the inside diameter measurements are given in Table 6. The results show that Dia. B is 0.002 out of round and 0.0005 over size at one location. Dia. C is only out of round by 0.0005 and within tolerance at all locations measured.

Table 6 - Bearing I. D., P/N H109-0039, Inspection Results

A.) Drawing dimension: 16.9995 - 17.0020 in.

Dia C Loc	Dimension (in.)	Dia B Loc	Dimension (in.)
1-2	17.0005	1-2	17.0005
3-4	17.0010	3-4	17.0015
5-6	17.0010	5-6	17.0020
A-B	17.0005	A-B	17.0025
C-D	17.0005	C-D	17.0015
E-F	17.0005	E-F	17.0005

The slider/bearing assembly was fit checked with the cylinder at room temperature. No interference was encountered and the bearings did not bind. In fact, with only one bearing over the cylinder and the slider held at a considerable angle, the one bearing did not bind but slid freely over the cylinder.

To verify the thermal expansion of the bearing inside diameters while constrained in the slider, the assembly was instrumented with the same nine thermocouples in the same locations as described previously in the thermal expansion test procedure. The bearing inside diameter expansion was measured at 300 and 400 °F. The overall average coefficient of thermal expansion along with its 95% confidence interval is given below.

Slider/Bearing Assembly:

$$\alpha = 7.416\text{x}10^{-6} \pm 0.157\text{x}10^{-6} \, \frac{1}{°\text{F}}, \text{ where } n=24$$

This is less than the predicted coefficient from Equation 10, but is still larger than the measured coefficient for the cylinder. Using the measured dimensions and thermal expansion coefficients for the cylinder and slider/bearing assembly, a bearing clearance analysis was conducted for the range of temperatures at which the main valve may run.

- Bearing clearance at room temperature: +0.0055 to +0.0070 in. (Dia.)
- Bearing clearance at 570 °F: +0.0072 - +0.0087 in. (Dia.)

These values were determined from the estimated mean thermal expansion coefficients. To consider the worse possible case, we take the lowest possible slider/bearing coefficient from its 95% confidence interval, and the highest possible cylinder coefficient from its confidence interval and calculate the bearing clearance at high temperature.

- Smallest possible bearing clearance, 570 °F: +0.0048 to +0.0063 in (Dia.)

This is still an adequate clearance and exceeds the 0.0040 minimum design clearance. All of the rebuilt main valve test results indicate that the identified causes for the main valve failure have been addressed and fixed. The cylinder and slider/bearing assembly, along with the excess bearing material, were shipped back to WPAFB on April 26, 1993. As indicated in a memo from Dunn and Haldeman on April 18, the following tasks should be completed upon the arrival of the main valve.

Main Valve Tasks at WPAFB

1.) Install the main valve into ATARR.

2.) Adjust the valve to obtain vacuum integrity.

3.) Cycle the valve several times at room temperature while recording its position. Look for changes in the time to open and close and check for wear marks on the matting surfaces. Either one may be an indication of frictional problems. Note that the LEDS must be installed on the downstream side of the valve for these tests.

4.) Heat the supply tank to approximately 425 °F while keeping the pressure at 1 atm.. Allow the system to soak for several hours and repeat task 3.

5.0 SUMMARY

The possible causes for the Dec. 16th main valve failure were identified and verified to the extent possible. The soft nickel platting on the cylinder was deemed inappropriate in this application from the evidence of its scoring before and after the failure. The cylinder was reworked by replacing the nickel with a hard chromium platting. The previous bearing configuration was exposed as an unnecessary risk to binding the slider on the cylinder and the problem was remedied by changing the bearing face from a flat surface to a large radius. The last and most critical problem identified was the incompatibility of the thermal expansion rates of the initial valve components. The extent of this problem was determined by a comprehensive experimental investigation aimed at measuring the thermal expansion rate of each valve component. It was only through this investigation that it was possible to choose a bearing material to alleviate the thermal expansion problem.